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No. 572

PERFORMANCE OF AIR-COOLED ENGINE CYLINDERS
USING BLOWER COOLING

By Oscar W. Schey and Herman H. Ellerbrock, Jr.
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SUMMARY

An investigation was made to obtain information on the minimum quantity of air and power required to cool conventional air-cooled cylinders at various operating conditions when using a blower. A Pratt and Whitney Wasp H and a Pratt & Whitney Wasp D cylinder were used with two different cooling-air jackets on each cylinder. The cooling air was supplied by a blower and measured with a Durley orifice box. Cylinder temperatures were measured with iron-constantan thermocouples connected to a portable pyrometer. Tests were made at engine speeds varying from 1,500 to 2,100 r.p.m. with atmospheric pressure at the carburetor and at an engine speed of 1,900 r.p.m. with carburetor-intake pressures varying from 20 to 35 inches of Hg absolute. The weight of the cooling air was varied from approximately 0.56 to 1.82 pounds per second.

The results of these tests showed that with blower cooling 1.09 pounds of air per second properly directed were sufficient to cool satisfactorily the Wasp H cylinder when it was developing 45 brake horsepower. Approximately 2 to 6 percent of the brake horsepower of the engine was required for satisfactory cooling, depending on the engine operating conditions and the cylinder and jacket combination used. The power required for cooling varied as the 2.81 to 2.90 power of the cooling-air weight. The temperature difference between the cylinder and the cooling air varied inversely as the 0.4 to 0.6 power of the weight of the cooling air, depending on the location of the thermocouple. The total head drop across the cylinder varied as the 1.77 to 2.04 power of the cooling-air weight. The shape of the jacket had a large effect on the temperature distribution; increasing the velocity over the front of the cylinder reduced the temperatures of the whole cylinder.

INTRODUCTION

Many researches have been conducted during the last few years on methods for reducing the drag and improving the cooling of radial air-cooled engines. The N.A.C.A. cowlings have been used in practically all of these investigations. The effect of the use of baffles and trailing-edge flaps to limit, direct, and control the quantity of cooling air when N.A.C.A. cowlings are used has recently been investigated. These investigations have resulted in a large reduction of drag and a large improvement in cooling (reference 1).

The use of controllable propellers to maintain constant power output for all operating conditions has made the cooling problem more difficult because the engine power in climb may be the same as in level flight, whereas the air speed in climb is about one-half that in level flight. The difference in slipstream velocity is less than the difference in air speed but there would, nevertheless, be a large difference in temperature for the two conditions. The cooling system should be so designed that the degree of cooling obtained would have a definite relation to the power developed in order that practically uniform temperatures may be obtained for all conditions at or near full throttle. A blower geared to the engine has been suggested as a means of controlling and improving the cooling. An important advantage of blower cooling is that the air-cooled, like the liquid-cooled, engine will be self-contained and consequently more independent of the engine installation.

The purpose of the present research was to investigate blower cooling of air-cooled engine cylinders. The quantity of air and the blower power required for cooling a single-cylinder air-cooled engine were determined. The tests were conducted on two cylinders having fins of different design.

APPARATUS

The apparatus consisted of a single-cylinder air-cooled engine, a supercharger for boosting carburetor-intake pressures, an electric dynamometer, a cooling system, and the necessary instruments to measure the factors involved. A

diagrammatic sketch of the set-up is shown in figure 1 and a photograph of the engine with the cylinder enclosed in the cooling jacket is shown in figure 2.

Air-cooled cylinders.— The two air-cooled cylinders (fig. 3) used in these tests were from Pratt & Whitney Wasp D and H engines. They were adapted to a universal test-engine base (reference 2) to operate with a stroke of 6 inches and at compression ratios of 5.39 and 5.33 for the D and H cylinders, respectively. The fins on the head of cylinder H, which is of a later design than cylinder D, are more closely spaced than are those on the head of cylinder D. The fins on the barrels of the two cylinders are of the same pitch but the width is different, those for the Wasp D being $3/8$ inch wide and those for the Wasp H being $1/2$ inch wide. Each cylinder was provided with an oil feed line to supply oil to the piston skirt through four 0.067-inch-diameter holes drilled in the cylinder wall near the base.

Cylinder jackets.— The cylinders were enclosed in sheet-metal jackets fitting close to the fins and open at the front and the rear (figs. 4 and 5). Two jackets were tested on each cylinder, jackets A and B on cylinder D and jackets C and D on cylinder H. Jackets A and C had entrance passages of small cross-sectional area, the air passing between the valve rods; jackets B and D enclosed the valve rods and allowed the air to pass over them. Jacket C was made by placing inserts in the entrance of jacket D to direct the air so that it passed between the valve rods. Care was exercised in the design of the entrance of the jackets to insure no breakaway of the air from the walls. The ratio of the area of the exit of the jacket to the clear area between the fins for cylinder D was 2.6 for the barrel fins and 1.8 for the head fins. The jackets for cylinder H were designed so that the exit area was twice the clear area between the fins because information obtained from tests with baffles on electrically heated cylinders showed that the ratio of the area of the exit between the baffles to the clear area between the fins should be approximately 2 for the highest over-all surface heat-transfer coefficient (reference 3). The information on electrically heated cylinders was not available for the design of jackets for cylinder D. The length of the exits on all the jackets was 6 inches.

Test equipment.- An N.A.C.A. Roots supercharger was used to increase the carburetor-intake pressure during the boost tests. A tank was placed in the air duct between the supercharger and the engine to eliminate pressure pulsations caused by these units.

An electric dynamometer was used to absorb the power and to measure the torque of the engine. A Stromberg NAL-5 carburetor modified by installing needle valves in the main jets was used on the test engine. A small weighing tank suspended from a sensitive balance that electrically operated a revolution counter was used to measure the fuel; the length of a fuel run was the time required to consume 1/2 pound of fuel.

The cooling-air system consisted of a blower to supply the cooling air, an orifice tank to measure the quantity of air, and an air duct between the blower and the jacket enclosing the cylinder. Baffles and screens were placed in the air duct to insure uniform air flow across the entrance of the jacket.

An electrically controlled stop watch and revolution counter were used to determine the engine speed.

Temperature and pressure measurements.- Iron-constantan thermocouples and a direct-reading portable pyrometer were used to measure the cylinder temperatures. The thermocouples were made from 0.016-inch diameter wire and were peened in the cylinder head and spot-welded to the barrel. The thermocouples on cylinders D and H were located as shown in figure 3. Chromel-constantan thermocouples were used to obtain the air temperatures at the inlet of the jacket. The cold junctions of all the thermocouples were placed in an insulated box. Alcohol thermometers were used to measure the temperature of the air entering the orifice tank, of the cold-junction box, and of the carburetor intake.

Water manometers were used to measure the pressure in the orifice tank and at the inlet of the jacket; a mercury manometer was used to measure the carburetor-intake pressure.

METHODS

Tests.— Tests were conducted with both cylinders at engine speeds from 1,500 to 2,100 r.p.m. with atmospheric pressure at the carburetor intake. At 1,900 r.p.m. with cylinder H, the carburetor-intake pressure was varied from 20 to 35 inches of mercury absolute in increments of approximately 5 inches of mercury. The range of cooling-air weight for the foregoing tests was from 0.56 to 1.82 pounds per second, which corresponded to air speeds of approximately 65 to 210 miles per hour on the Wasp D cylinder and of 90 to 285 miles per hour on the Wasp H cylinder, at an air temperature of 70° F. and a pressure of 29.92 inches of mercury. These air speeds are based on the free-flow area between the fins of the two cylinders. The fuel consumption was kept approximately constant at 0.55 pound per brake horsepower per hour. Observations were made of the engine torque, the engine speed, the fuel consumed, the carburetor-intake pressure and temperature, the spark setting, the temperature of the air entering the orifice tank, the temperatures of the air entering the jacket, the cylinder temperatures, the pressure drop across the orifice tank, the pressure at the entrance of the jacket, and the barometric pressure.

The weight of the cooling air was controlled by varying the speed of the blower. The carburetor-intake pressures were varied by either throttling the intake of the surge tank or boosting with the supercharger.

Gasoline conforming to Army Specification Y-3557-G and having an octane number of 87 was used for most of the tests. For the most severe conditions ethyl fluid was added to the fuel in a sufficient amount to suppress audible detonation.

Only temperatures for a few representative thermocouples are submitted in the report; the discussion of the results and the conclusions are, however, based on a study of all the cylinder temperatures.

Computations.— The engine horsepowers given in this report are all observed values and were calculated from the corrected dynamometer-scale reading and the engine

speed. No correction was applied to engine power for variation from standard conditions of barometric pressure, atmospheric temperature, or humidity because these corrections could not be applied with the same degree of accuracy to the cylinder temperatures. The method of computing the cooling-air weight is given in detail in reference 4. The power required for cooling was obtained by multiplying the total head at the entrance of the jacket by the air velocity and the area of the section where the total head is measured and includes both the loss across the cylinder and the kinetic energy of the discharge air.

The cylinder-temperature readings were corrected for instrument calibration, cold-junction temperature, and variation in inlet cooling-air temperature from 70° F. The latter correction was based on tests of several cylinders in which the temperature of the inlet air was varied; it was found that for every degree rise in air temperature, the head temperatures increased approximately 0.8° F., the barrel temperatures 0.7° F., the spark-plug-gasket temperature 0.84° F., and the flange temperatures 0.6° F. The inlet cooling-air temperatures were corrected for instrument calibration and cold-junction temperature.

The specific fuel consumption was calculated from the observed weight of fuel used, the time required to use this fuel, and the observed brake horsepower.

PRECISION

The cylinder temperatures are considered to be accurate to within $\pm 4^{\circ}$ F.

The temperatures of the inlet air are accurate to within $\pm 2^{\circ}$ F.

The pressure at the inlet of the jacket varied owing to fluctuations caused by the blower, but the results are accurate to within $\pm 1/8$ inch of water.

The manometer measuring the pressure drop across the

metering orifice fluctuated considerably when a large amount of air was delivered. The error is considered to be approximately ± 2.5 percent at cooling-air weights of 1.3 pounds per second and to decrease until at approximately 0.8 pound per second the error is less than 1 percent.

The torque-scale readings are correct to within ± 1 percent.

The accuracy of the fuel weights varied from 0 to $-2-1/2$ percent.

RESULTS AND DISCUSSION

Effect of the shape of the jacket on cooling.— The curves of figure 6 show that for the same weight of cooling air the cylinder temperatures were lower at all points with jacket A than with jacket B. As the rear half of the two jackets is the same, it can be concluded from these results that increasing the velocity of the cooling air over the front of the cylinder will also reduce the temperature at the rear of the cylinder. The difference in the temperatures obtained with the two jackets was greatest for the points that had a high temperature and for the points on the front part of the cylinder. Thermocouple 15, which was located below the front spark-plug boss, showed a maximum difference of 152° F.; whereas thermocouple 14, which was located on the flange and at the rear of the cylinder, showed a difference of from 0° to 5° F.

The shape of the jacket had a large effect on the temperature difference between points on the cylinder, particularly between points at the front and the rear. For example, with jacket B the temperatures for points below the front spark plug were appreciably higher (30° to 60° F) than those for points below the rear spark plug; with jacket A the temperatures below the rear spark plug were from 35° to 60° F. higher than those below the front plug. The temperatures at points above the spark plugs and on the barrel were higher in the rear of the cylinder than at the front regardless of the jacket used; with jacket A the maximum temperature difference was 120° F. for the barrel,

whereas with jacket B the maximum temperature difference was only 12° F. It might be well to point out that with either jacket the temperature differences between the front and the rear were never greater than 120° F. and this difference was confined to one point; in flight on installations using ring cowlings, temperature differences as high as 150° F. have been obtained (reference 5). A jacket of proper design should be of great assistance in reducing distortion caused by excessive temperature differences and should also be an excellent means for eliminating hot spots in the cylinder.

The difference in temperatures between the two jackets on cylinder H (fig. 7) was slight as compared with the temperature differences between the two jackets on cylinder D. As might be expected, however, the largest temperature difference between the two jackets was for the points on the front of the cylinder; for thermocouple 2 the difference was 26° F. and for thermocouple 15 the difference was 44° F. The slight difference between the shape of the two jackets used on this cylinder (fig. 5), and probably the better finning, explains the small difference in temperatures obtained.

In general, it may be said that the temperature distribution of cylinder H was similar to that obtained on cylinder D. With jacket D on cylinder H the rear temperatures were higher than the front except for points a short distance below the spark plugs. The maximum temperature difference, 69° F., between the front and rear of cylinder H was on the barrel, in comparison with the maximum temperature difference of 120° F. on the barrel of cylinder D. The greater amount of finning on cylinder H tended to reduce the temperature difference and the number of hot spots.

Effect of the cooling-air weight on the difference between cylinder and air temperatures.— The test results for cylinder D show that the temperature difference between the cylinder and the cooling air (temperatures in fig. 6 less 70° F.) varied inversely as the 0.45 to 0.60 power of the air weight depending upon the location of the thermocouples. As the wall-temperature differences are inversely proportional to the wall heat-transfer coefficients for a given heat output, an examination of

the theoretical equation for the wall heat-transfer coefficient will provide an insight into the reasons for the variation in the exponent noted.

The following equation from reference 6

$$U = \frac{q}{s+t} \left[\frac{2}{a} \left(1 + \frac{w}{2R_b} \right) \tanh aw' + s_b \right] \quad (13)$$

gives the wall heat-transfer coefficient U as a function of fin dimensions, conductivity, and fin surface heat-

transfer coefficient. In this equation $a = \sqrt{\frac{2q}{kt}}$, q is

the surface heat-transfer coefficient of the cylinder, k is the conductivity of the metal, t is the thickness of the fins, and w' is the width of the fins plus half the fin thickness. Reference 6 also shows that the fin surface heat-transfer coefficient q varies as $(V\rho g)^n$. It is evident from equation (13) that for small values of

$\sqrt{\frac{2q}{kt}} w'$, U varies as $(V\rho g)^n$ and that as $\sqrt{\frac{2q}{kt}} w'$ increases,

the exponent approaches asymptotically the value $n/2$. For engine cylinders the value of the exponent lies, in general, between 0.4 and 0.6.

In order to illustrate the foregoing discussion, the temperature differences between the average of 8 thermocouples on the barrel and the cooling air are shown in figure 8, together with the reciprocal of the wall heat-transfer coefficient U in B.t.u. per square inch per degree Fahrenheit per hour calculated from equation (13) for an electrically heated finned cylinder of the same dimensions. It will be noticed that in the same range of weight velocity of the cooling air the slopes of the two curves are almost equal. The weight velocity of the cooling air was obtained by dividing the cooling-air weight by the free-flow area between the fins. The weight velocity of the cooling air at the section considered is probably a little different from the average weight velocity of the air but, as figure 8 shows, the slope of the curve changes very little with a rather large change of weight velocity. The rela-

tion between cylinder temperature and air weight for cylinder H when plotted on logarithmic coordinate paper (see fig. 8 for cylinder D) indicates that the temperature difference between the cylinder and the cooling air varied inversely as the 0.40 to 0.50 power of the air weight.

Effect of cooling-air weight on total head drop across the jacket.— The curves of figure 9 show that the total head drop across the cylinder with different jacket and cylinder combinations varied from approximately 2.5 to 20 inches of water. These total heads include the drop across the cylinder and the kinetic energy loss at the exit of the jacket. The engine conditions for these curves were the same as for figures 6 and 7.

As the engine power increases, more heat is given to the cooling air, resulting in a change in air density. With the same weight velocity of the cooling air this change in density results in an increase in pressure drop. The change in total head drop across the cylinder due to density change, however, is so small that all points fall on the same curve.

The average slope of the curves in figure 9 is 1.88. Dryden and Kuethe (reference 7) have shown that for pipes and flat plates the drag is theoretically proportional to the 1.8 power of the velocity. Unpublished tests made at Massachusetts Institute of Technology by R. H. Smith and R. T. Sauerwein show that for various finned plates the drag varied as the velocity to the 1.75 to 1.96 power, depending on the pitch and width of the fins.

The closely spaced fins of cylinder H gave a much larger pressure drop than the widely spaced fins of cylinder D. There was no difference in pressure drop between jackets C and D on cylinder H, but the difference between the two jackets on cylinder D was appreciable for a given air weight. The entrances of the jackets on cylinder H were more nearly alike than were those on cylinder D and, furthermore, the closely spaced fins on cylinder H affected the pressure drop more than did the widely spaced fins on cylinder D.

Effect of cooling-air weight on the brake horsepower developed.— The curves in figure 10 show that increasing the quantity of cooling air had little effect on the brake

horsepower developed. As there is only a small gain in engine power when the engine is overcooled, only sufficient air should be provided to insure the reliability and the life of the engine.

Effect of cooling-air weight on the blower power required.- The curves in figure 6 show that the cooling of cylinder D was considerably better with jacket A than with jacket B. The curves in figure 11 show that to obtain this better cooling it was necessary to expend approximately twice as much power to force a definite weight of cooling air through jacket A than was required with jacket B. These curves also show that the difference in power required by the two jackets on cylinder H, like the difference in temperatures shown in figure 7, was small for a given air weight. The engine conditions for this figure were the same as for figures 6 and 7.

As the power required for supplying the air is roughly proportional to the product of the total head drop and the velocity of the air and as the total head drop varied as the 1.88 power of the air weight (fig. 9), the exponent of the power curves should be approximately 2.88. An average value of the curves in figure 11 shows that the power required for forced cooling varied as the 2.83 power of the air weight.

Minimum air weight and power required for satisfactory cooling.- The results submitted thus far have shown the power required to force the air past the cylinders and the relation between cylinder temperatures and air weight. In a selection of the most desirable jacket both factors must be considered. The cylinder and jacket having a combination of air weight and pressure drop giving satisfactory cooling with the minimum expenditure of energy is the most desirable.

Table I presents the comparative performance for each cylinder and jacket combination. The minimum power required to supply the cooling air with each combination is based on the power required to supply sufficient cooling air to maintain a temperature of 475° F. as indicated by thermocouple 29 located between the exhaust valve and the rear spark-plug boss. In the examination of the results submitted in this table it should be realized that the jackets do not necessarily represent the best design, for very little information was available on the design of the

jackets. With some refinement the losses through the jackets could probably be reduced without any sacrifice in cooling.

The table shows that from a power consideration cylinder D with jacket A is slightly better than cylinder H with jacket C. Although the weight of the cooling air that was supplied to cylinder H was less than that supplied to cylinder D, the pressure drop across cylinder H was greater and as a result the power requirements were slightly higher for cylinder H. That cylinder D should show better results than cylinder H was surprising and contrary to expectation because cylinder H had more fin area.

Table II shows the free-flow areas between the fins of cylinders D and H and the quantity of air required for satisfactory cooling of cylinder D with jacket A and of cylinder H with jacket C at an engine speed of 1,900 r.p.m. If the quantities of air are divided proportionally to the free-flow areas of the heads and barrels of cylinders D and H, then the head of cylinder D would get 13.8 cubic feet of air per second; the barrel of D, 3.5 cubic feet per second; the head of H, 9.0 cubic feet per second; and the barrel of H, 5.5 cubic feet per second, as shown in table II.

Although the difference in the lengths of the paths that the air followed around the head and around the barrel was neglected in the computation, the values are a good indication of how the air was divided between the head and the barrel for the two cylinders. With this proportion of air, the average velocity, based on the free-flow area, for cylinder D was approximately 150 miles per hour and for cylinder H approximately 170 miles per hour.

Decreasing the quantity of cooling air results in an increase in the rise of the cooling-air temperature across the cylinder and a consequent increase in cylinder temperatures. When 1 pound of air per second is used in cooling an air-cooled cylinder developing 45 brake horsepower, the temperature of the cooling air will increase 60° F. while passing from the front to the rear of the cylinder. As approximately 50 percent more air passed over the head of cylinder D than over the head of cylinder H, the heating of the cooling air of cylinder D should be considerably less and the head temperatures of cylinder D should increase less from this heating.

That the barrel temperatures for cylinder H are slightly higher than those for cylinder D must be attributed to causes other than the difference in finning or air flow because there was more finning on the barrel of cylinder H and, furthermore, the velocity of the cooling air was approximately 20 miles per hour higher for cylinder H. In these tests care was taken to maintain the same pressure on the oil feed line near the base of the cylinder since previous tests had shown that the quantity of oil delivered to the piston skirt had an appreciable effect on cylinder temperatures. Increasing the oil pressure on each cylinder from 1 to 8 pounds per square inch resulted in an average decrease in temperature for cylinders D and H of 10° F. and 30° F., respectively.

The results indicate that, in order to obtain satisfactory cooling with minimum power, the quantity of cooling air must be judiciously restricted, especially where low operating temperatures are desired. If the quantity of cooling air is greatly reduced, the temperature rise of the air in passing over the cylinder will increase and thus reduce the temperature difference between the cooling air and the cylinder, which will impair the cooling unless the fin surface is modified so that the velocity of the cooling air is increased.

The importance of the jacket shape can be appreciated from a comparison of the results obtained from jackets A and B on cylinder D. Although the engine power was higher and the air weight less, the average temperatures obtained with jacket A were lower than those obtained with jacket B. (See table I.) Furthermore, the power expended in cooling with jacket A was about 50 percent of that for jacket B.

The power required to force the cooling air past the cylinders for adequate cooling varied from approximately 2 to 6 percent of the engine power, depending on cylinder and jacket design and engine-operating conditions. On the assumption of an over-all blower efficiency of 65 percent, approximately 3.0 to 9.0 percent of the engine power should be required for cooling. Wood (reference 8) found that the power required to cool a cylinder with a fan was 4 percent of the engine output, including fan losses. In a discussion of Fedden's paper on air-cooled engines (reference 9), F. M. Green states that he has found from tests of a single cylinder that the power required for cooling was approximately 4 or 5 percent of the total power. Löhner (refer-

ence 10) gives a value of 3.5 percent of the brake horsepower required for cooling a multicylinder engine with blowers and 8.3 percent for a single-cylinder engine.

Effect of indicated horsepower on cylinder-temperature difference.— The curves in figure 12 show that the temperature difference between the cooling air and the cylinder varies as the 0.51 to 0.56 power of the indicated horsepower of the engine. Changes in the indicated power were obtained by varying the manifold pressures from approximately 20 to 35 inches of Hg absolute. Recent tests of a 2-row radial engine (reference 11) showed that the temperature difference varied as the 0.25 power of the indicated horsepower. In the tests of the 2-row radial engine the tunnel speed was held constant and the engine speed was held constant by varying the propeller pitch simultaneously with a change in manifold pressure. Apparently when the power was increased, the weight velocity past the cylinder increased because the slipstream air speed was greater at the higher power output than at the low and because the cooling air was more turbulent on account of the shape of the cowl and the disturbance created by the propeller.

GENERAL CONSIDERATIONS

The results submitted have shown that the difference in temperature between the cylinder and the air varied inversely as the 0.4 to the 0.6 power of the cooling-air weight and that the power required for cooling with each jacket varied approximately as the 2.83 power of the air weight. From these results it follows that, to cool an engine with minimum expenditure of power, an increase in cooling surface is to be preferred to an increase in air speed. The increase in cooling surface must be obtained by increasing the fin width and the number of fins from the consideration that with a definite air weight it is better to cool with a low than with a high velocity. The extent to which each may be increased is limited. The width is limited by the effectiveness of the fin, which is dependent on the thickness of fin and the thermal conductivity of the material; the spacing, aside from construction difficulties, is limited by the thickness of the fin and the amount that the space between the fins can be reduced without an appreciable decrease in air speed between the fins,

change in the nature of the flow, or excessive heating of the cooling air. On the basis of other tests conducted by this laboratory, an appreciable amount of cooling surface can be added before reaching limits of either heat flow or air flow. The excessive heating of air between closely spaced fins impairs cooling to such an extent that it is a major consideration for conditions with restricted air flow.

The maximum temperature specified for satisfactory cooling has a large effect on the blower power required. In figure 13 are plotted curves of percentage brake horsepower required for cooling at various temperature differences between cylinder and air for the four cylinder and jacket combinations at an engine speed of 1,500 r.p.m. The results from the four thermocouples that are plotted were obtained from figures 6, 7, 10, and 11. The curves indicate that it would be very desirable from a consideration of the power required for cooling to have an engine that could operate at high temperatures with no impairment of reliability or performance.

Tests are now in progress of a large number of service-type cylinders to study cooling under various operating conditions. From the tests of these cylinders and further tests with other jackets, more complete information about the shape of the flow passages will be obtained. There are also possibilities that the blower may be used to serve a twofold purpose: the removing of boundary layer from the airplane wing, and the supplying of cooling air to the engine.

CONCLUSIONS

The results of these tests show that:

1. The minimum power required for satisfactory cooling with an over-all blower efficiency of 100 percent varied from 2 to 6 percent of the engine power depending on the operating conditions.
2. The shape of the jacket had a large effect on the cylinder temperatures. Increasing the air speed over the front of the cylinder by keeping the greater part of the circumference of the cylinder covered by the jacket reduced the temperatures over the entire cylinder.

3. The temperature difference between the cylinder and the cooling air varied inversely as the 0.4 to 0.6 power of the cooling-air weight depending on the location of the thermocouple; those on the barrel varied as the higher power of the air flow.

4. The total head drop across the cylinder varied as the 1.77 to 2.04 power of the cooling-air weight depending on the cylinder and jacket combination. The power required for cooling varied as the 2.81 power of the cooling-air weight for three of the cylinder and jacket combinations and as the 2.90 power for the fourth combination.

5. An air quantity of 1.09 pounds per second properly directed and kept in contact with the cylinder would satisfactorily cool the Wasp H cylinder when it was developing 45 brake horsepower, the maximum temperature being 475° F.

Langley Memorial Aeronautical Laboratory,
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TABLE I

Comparison of Cylinders and Jackets for Satisfactory Cooling
(Carburetor-intake pressure, atmospheric)

Engine speed	Cylinder	Jack- et	Air weight	Total head drop across jacket	Brake horsepower	Friction horsepower	Barrel ring average ¹ temp.	Average ² head temp.	Barrel temp.		Head temp.		Power required for cooling ³	
									Min.	Max.	Min.	Max.		
r.p.m.			lb./sec.	in. water			°F.	°F.	°F.	°F.	°F.	°F.	hp.	Percent
1500	D	{A	0.96	5.3	37.1	6.2	270	362	229	345	223	486	0.77	2.08
		{B	1.40	7.4	34.6	6.3	271	383	230	314	258	487	1.35	3.90
	H	{C	.80	8.2	35.6	6.6	270	371	245	320	255	475	.82	2.33
		{D	.78	7.7	36.4	7.8	263	373	239	313	230	475	.76	2.09
1700	D	{A	1.05	6.4	39.8	8.2	263	347	223	326	207	492	.99	2.49
		{B	1.70	11.0	39.6	8.2	285	381	249	317	239	502	2.34	5.91
	H	{C	-	-	-	-	-	-	-	-	-	-	-	-
		{D	1.00	12.1	39.7	9.4	269	368	235	314	247	475	1.55	3.91
1900	D	{A	1.30	9.6	44.0	10.3	265	346	227	330	206	502	1.86	4.23
		{B	1.81	12.3	44.0	10.4	290	382	248	319	243	499	2.75	6.25
	H	{C	1.09	13.9	45.3	10.4	270	364	247	313	225	475	1.95	4.30
		{D	1.18	16.0	43.6	10.9	264	370	231	300	244	475	2.45	5.62
2100	H	D	1.26	18.2	47.7	12.4	265	365	231	320	237	475	2.97	6.22

¹Average of thermocouples 2-9, inclusive.

²Average of thermocouples 13 and 15-34, inclusive.

³Based on 100 percent blower efficiency.

TABLE II.

Comparative air flows over head and barrel of cylinder D
with jacket A and of cylinder H with jacket C

Cylinder	Jacket	Free flow area		Average air velocity	Air weight required (1900 r.p.m.)	Air flow over head ¹	Air flow over barrel ¹
		Head	Barrel				
		sq.in.	sq.in.	m.p.h.	lb./sec.	cu.ft./sec.	cu.ft./sec.
D	A	9.0	2.3	150	1.30	13.8	3.5
H	C	5.2	3.2	170	1.09	9.0	5.5

¹Based on air temperature 70° F., air pressure 29.92 in. Hg absolute.

$$C = .069$$

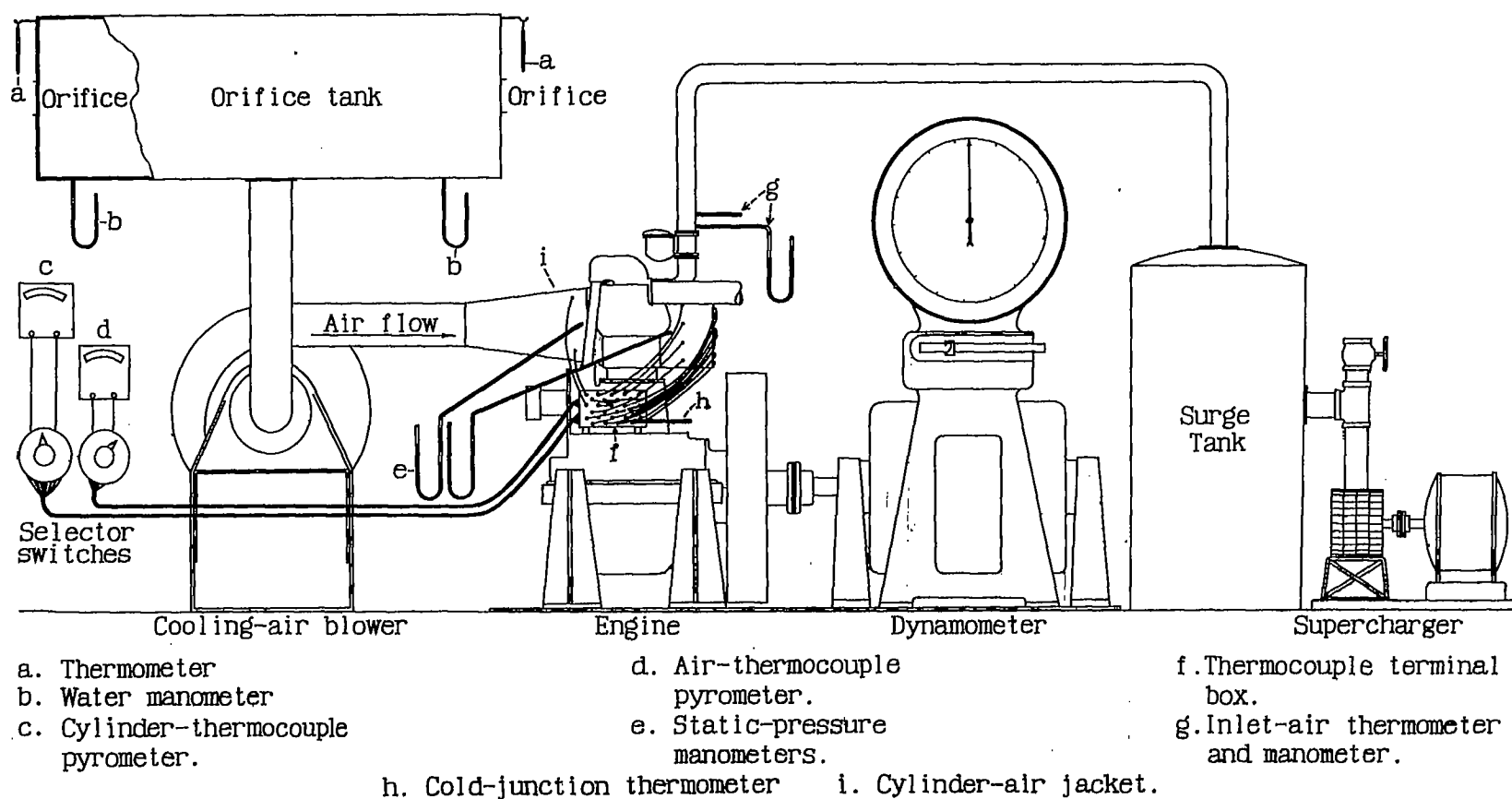


Figure 1.-Diagrammatic layout of equipment.

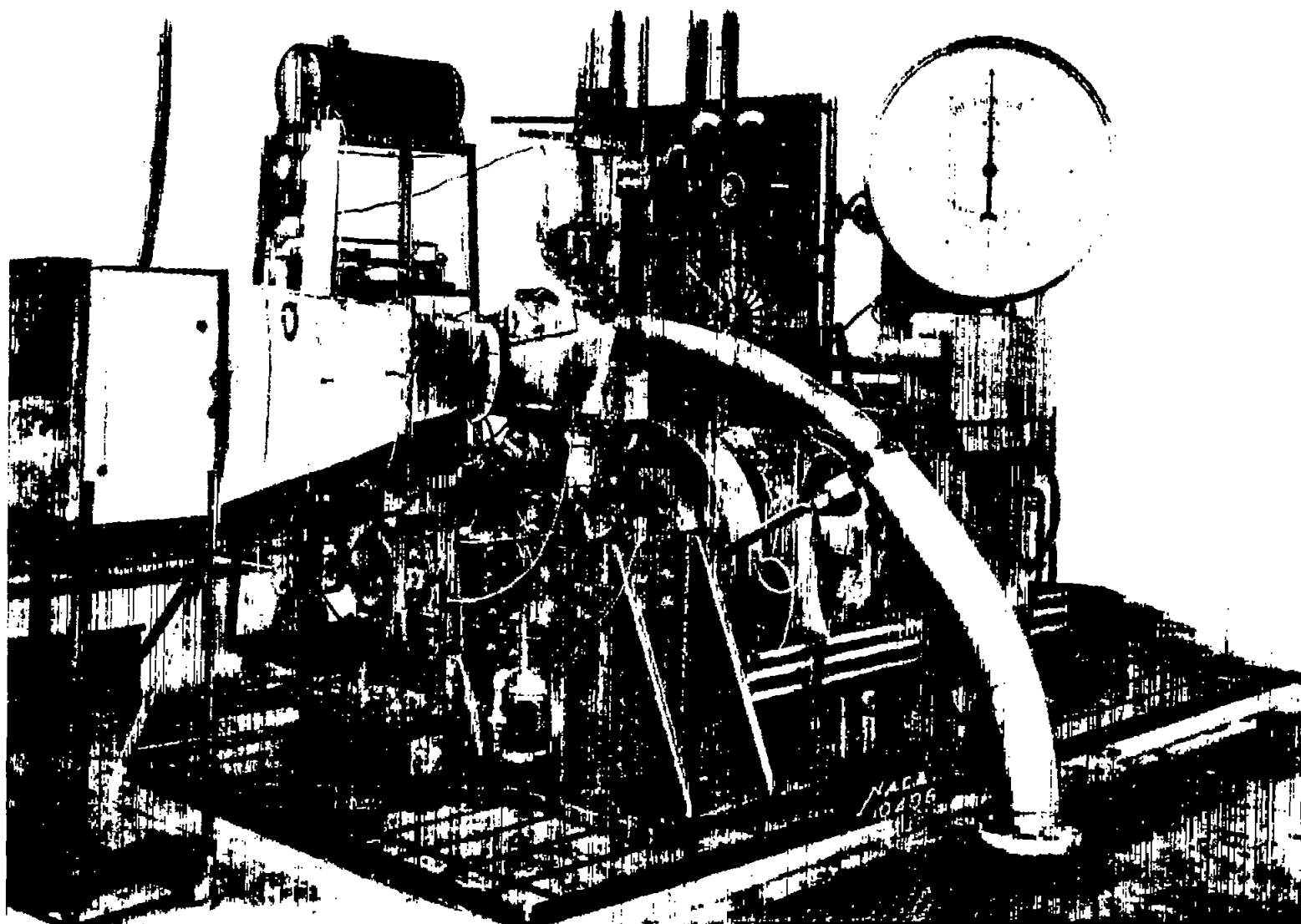
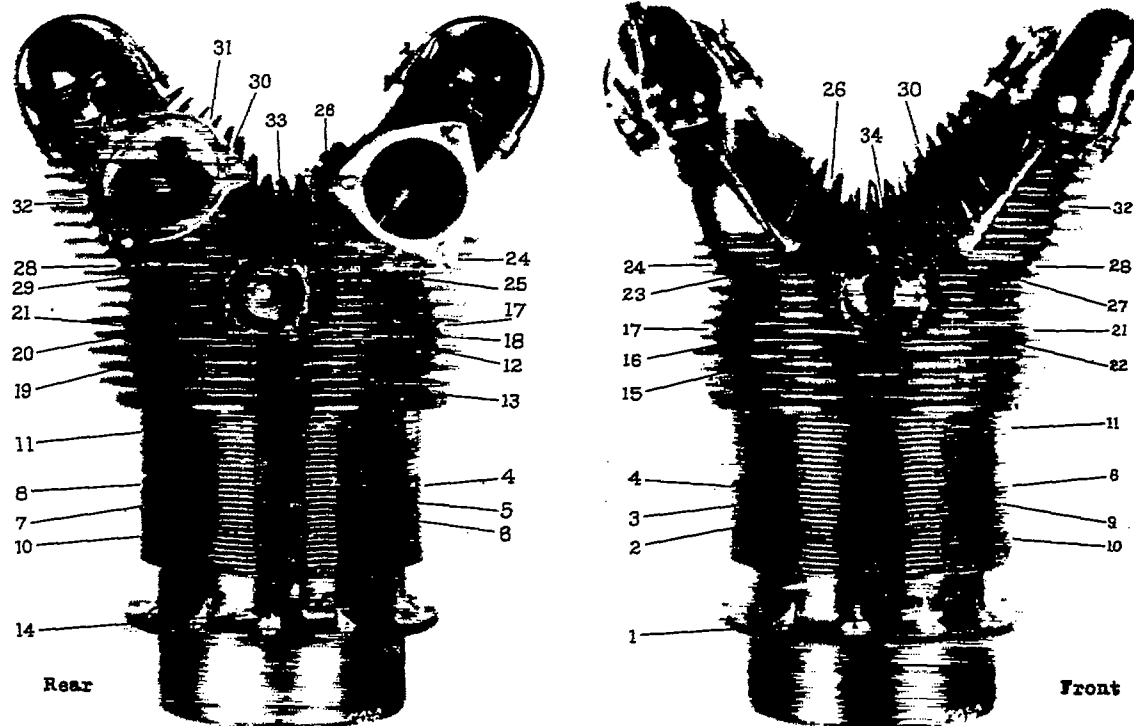
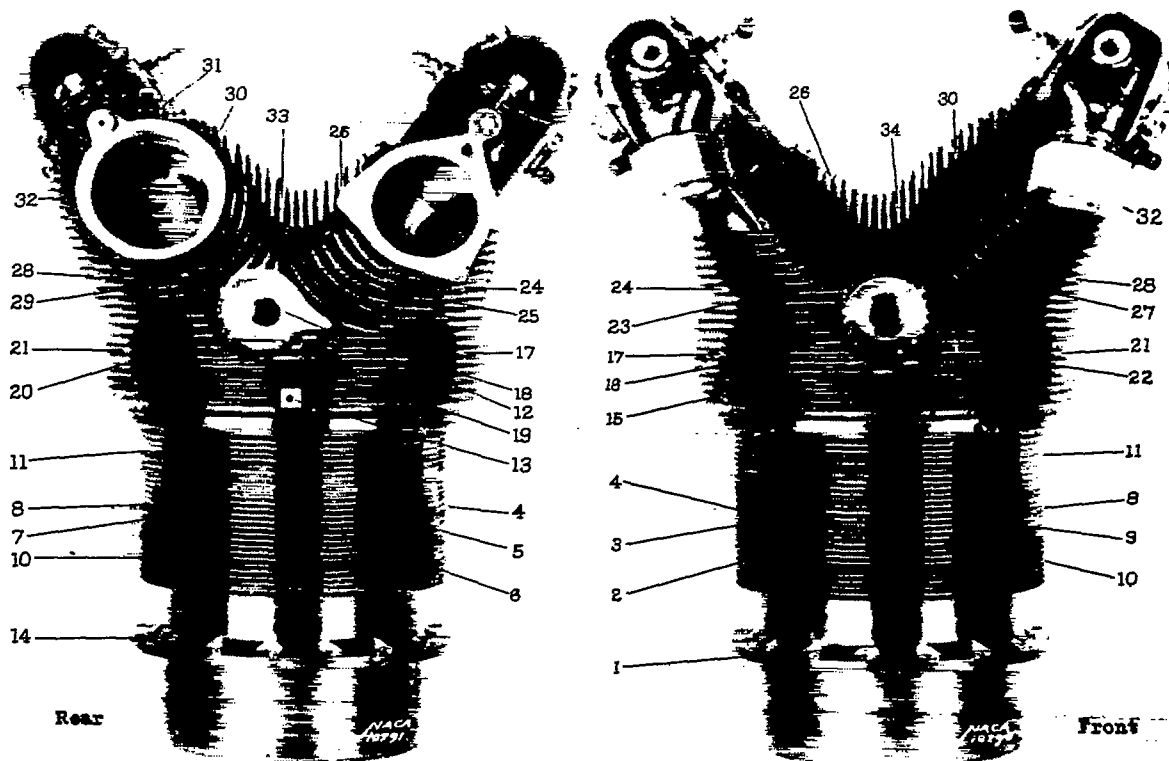


Fig. 2

Figure 2.- Set-up of single-cylinder air-cooled engine showing jacket and air duct.



(a) Wasp D cylinder. Thermocouple 13 is of spark plug gasket type



(b) Wasp H cylinder. Thermocouple 13 is of spark plug gasket type

Figure 3.- Front and rear views of cylinders showing location of thermocouples

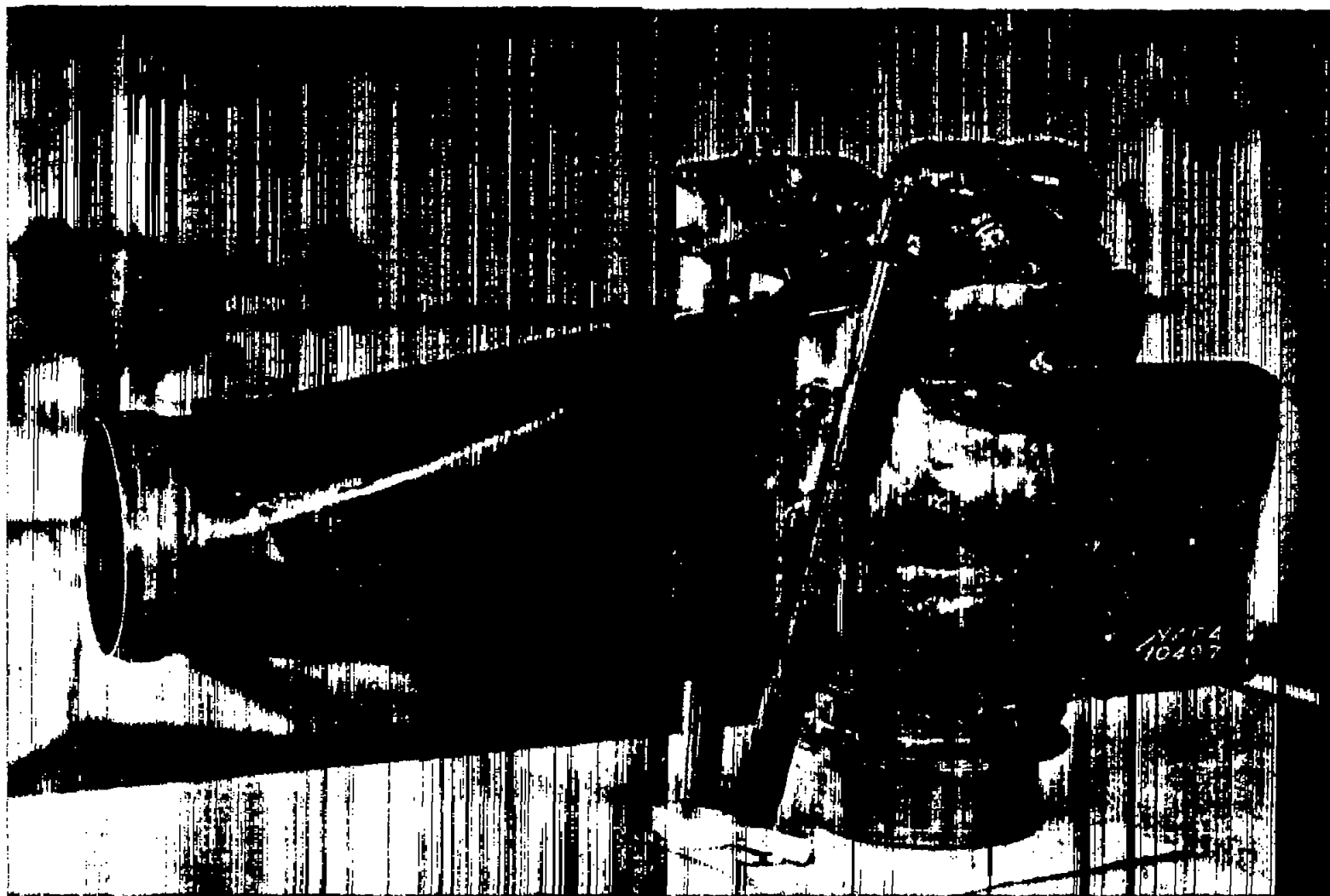


Figure 4.- Wasp D cylinder with jacket A in place.

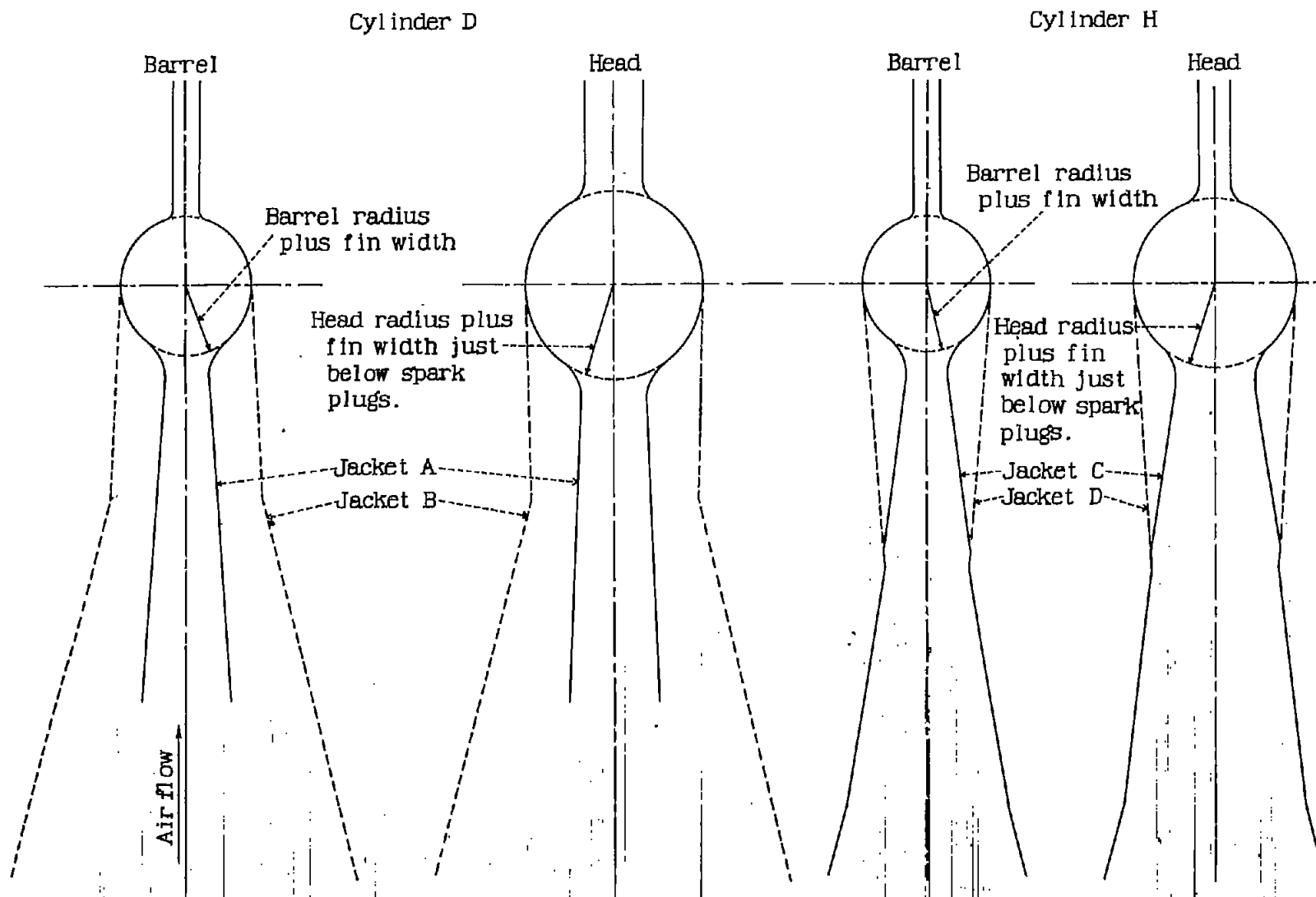


Figure 5.-Shape of jackets and degree of contact with cylinders.

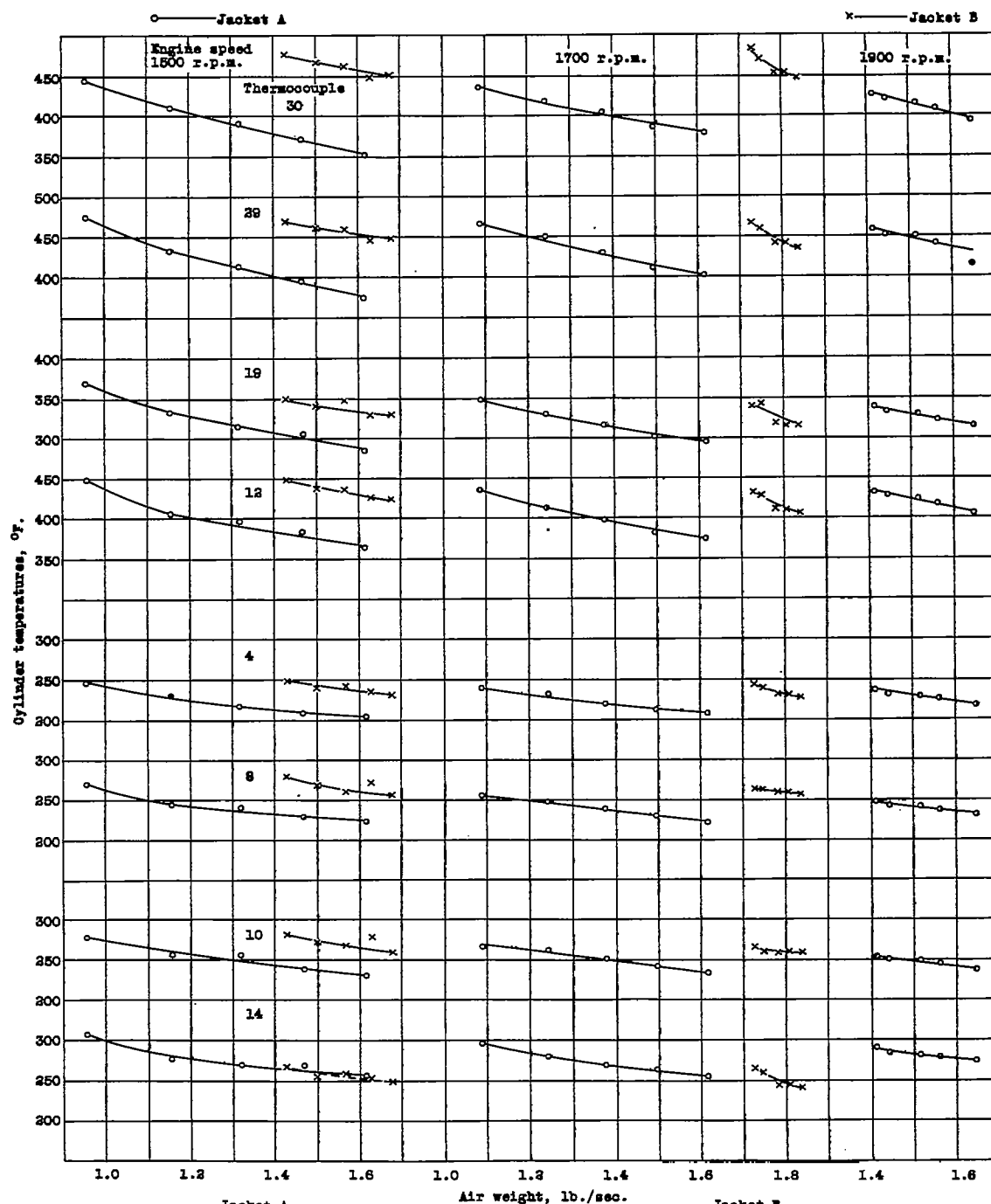


Figure 8.—Effect of cooling-air weight on temperatures of cylinder D with jacket A at three engine speeds and jacket B at two engine speeds.

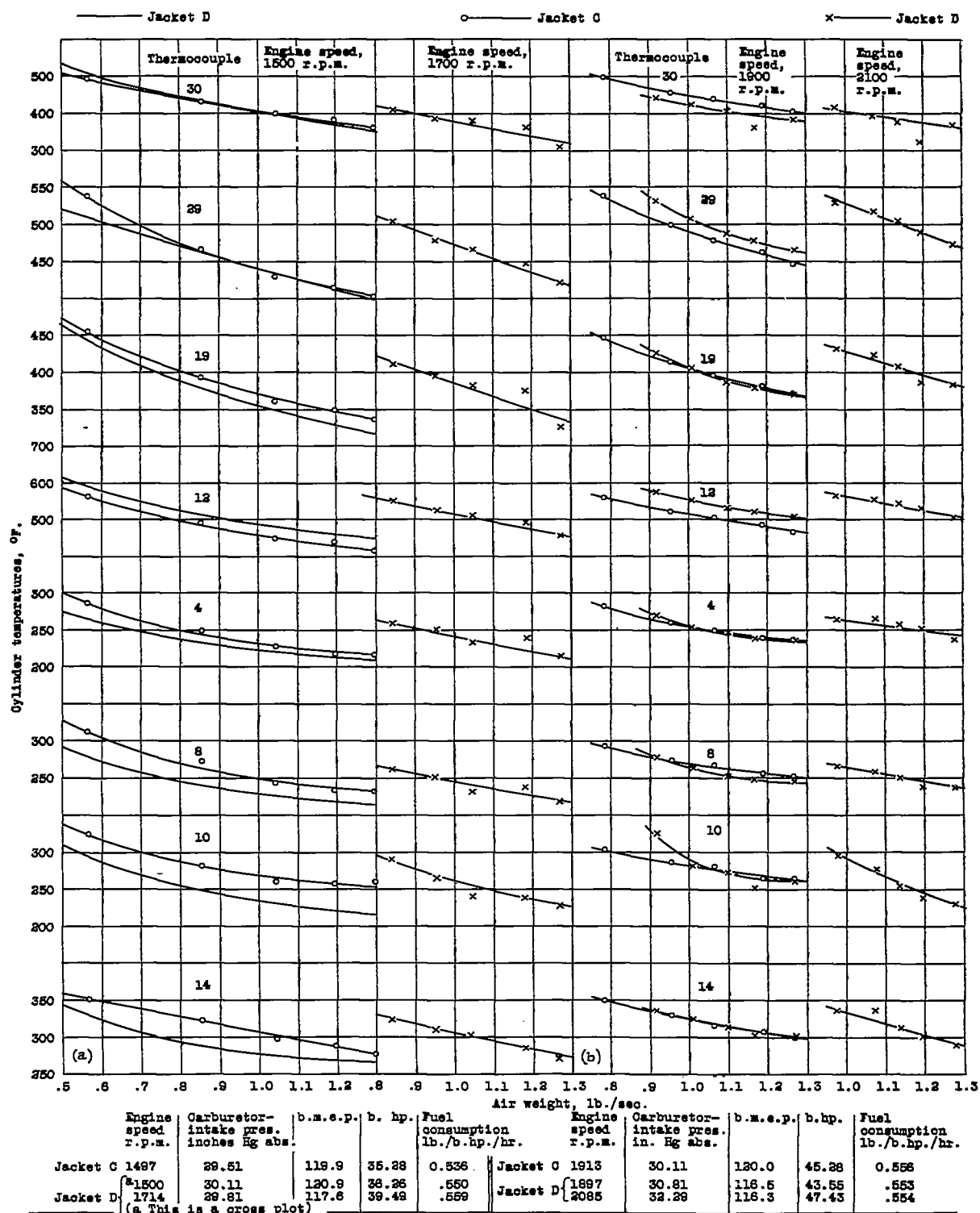


Figure 7a.—Effect of cooling-air weight on temperatures of cylinder H with jacket D at two engine speeds and jacket C at one engine speed.

Figure 7b.—Effect of cooling-air weight on temperatures of cylinder H with jacket D at two engine speeds and jacket C at one engine speed.

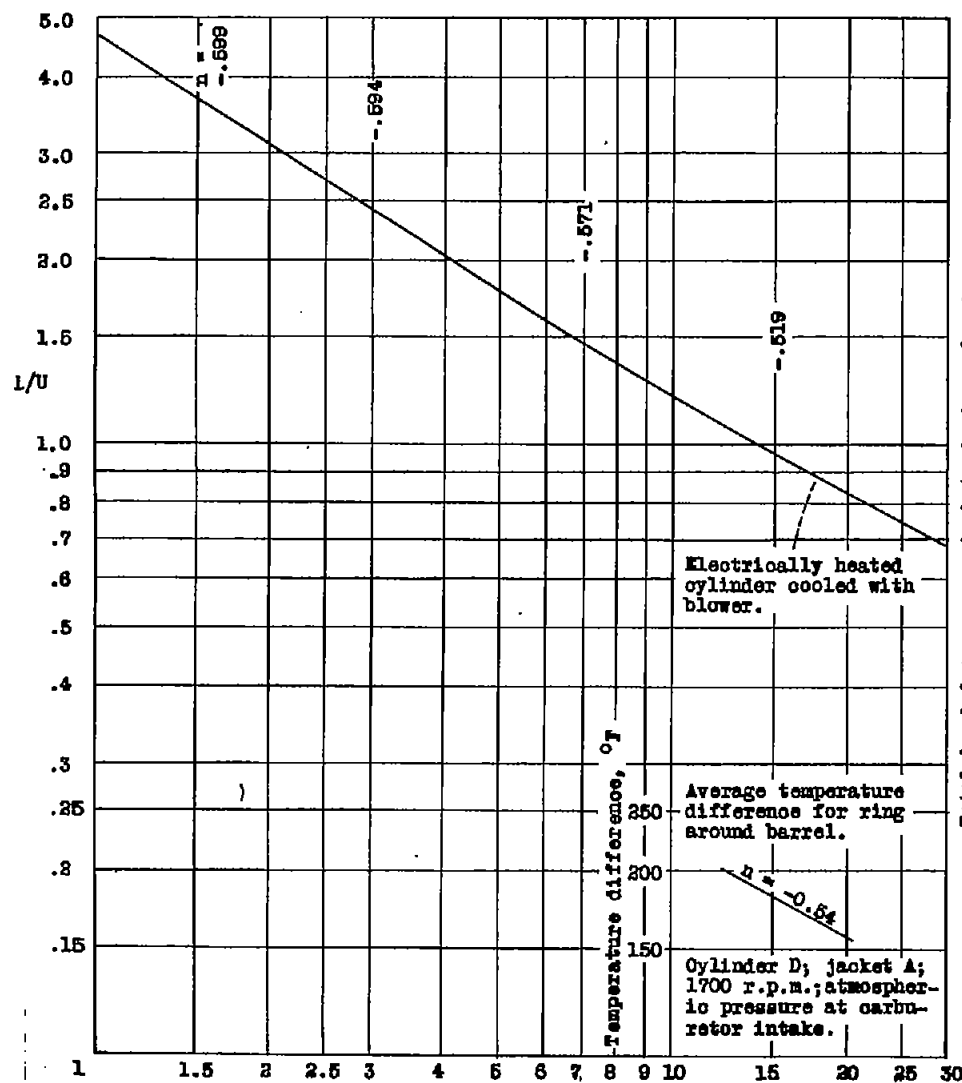


Figure 8.—Variation in wall heat-transfer coefficient U and temperature difference with weight velocity of cooling air.

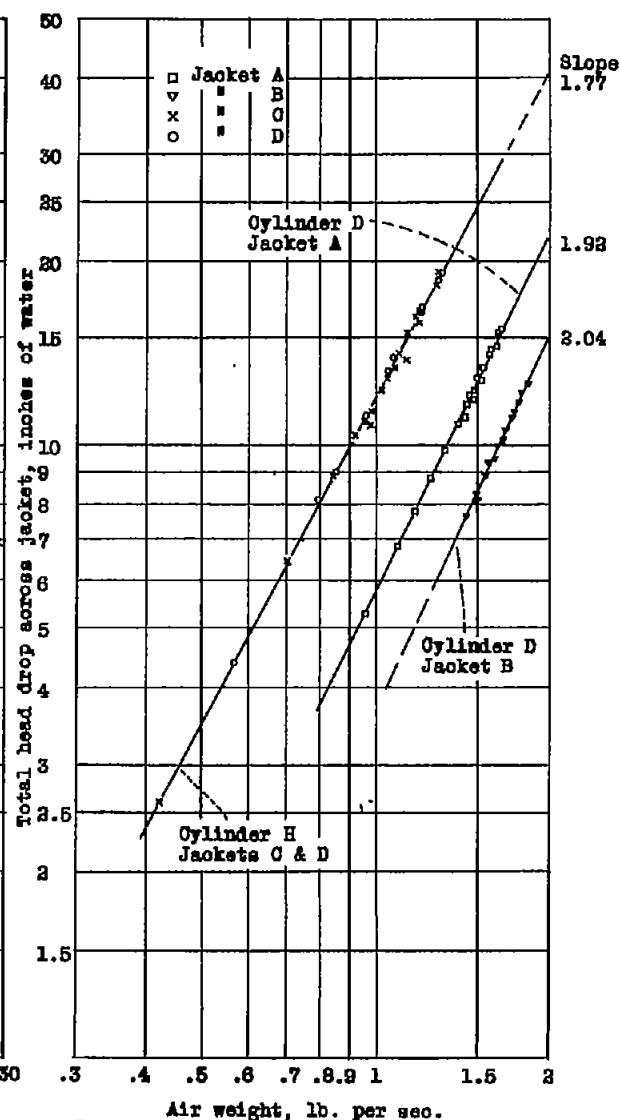


Figure 9.—Effect of cooling-air weight on total head drop across cylinders.

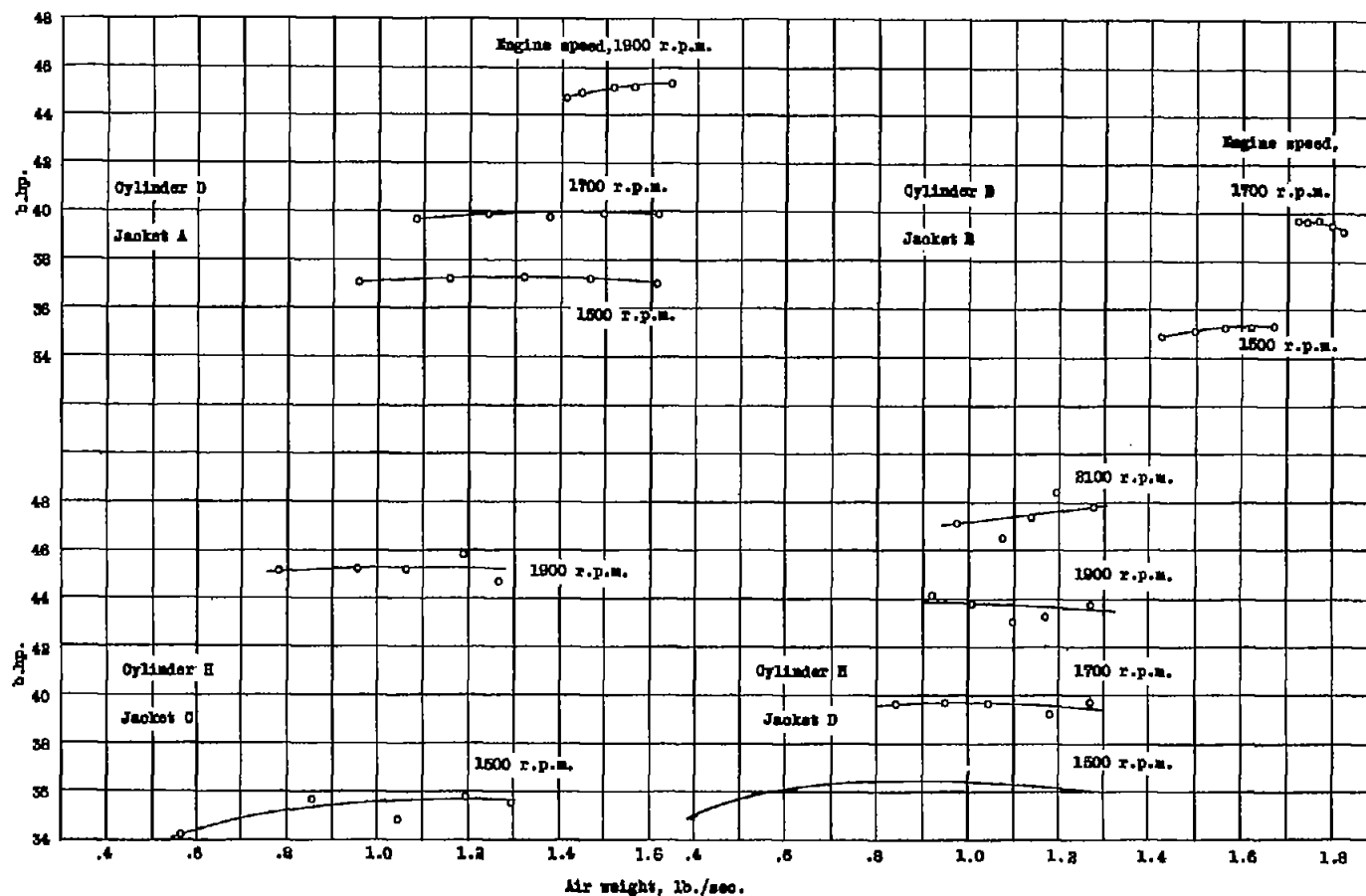


Figure 10.—Variation of brake horsepower with cooling-air weight at several engine speeds.

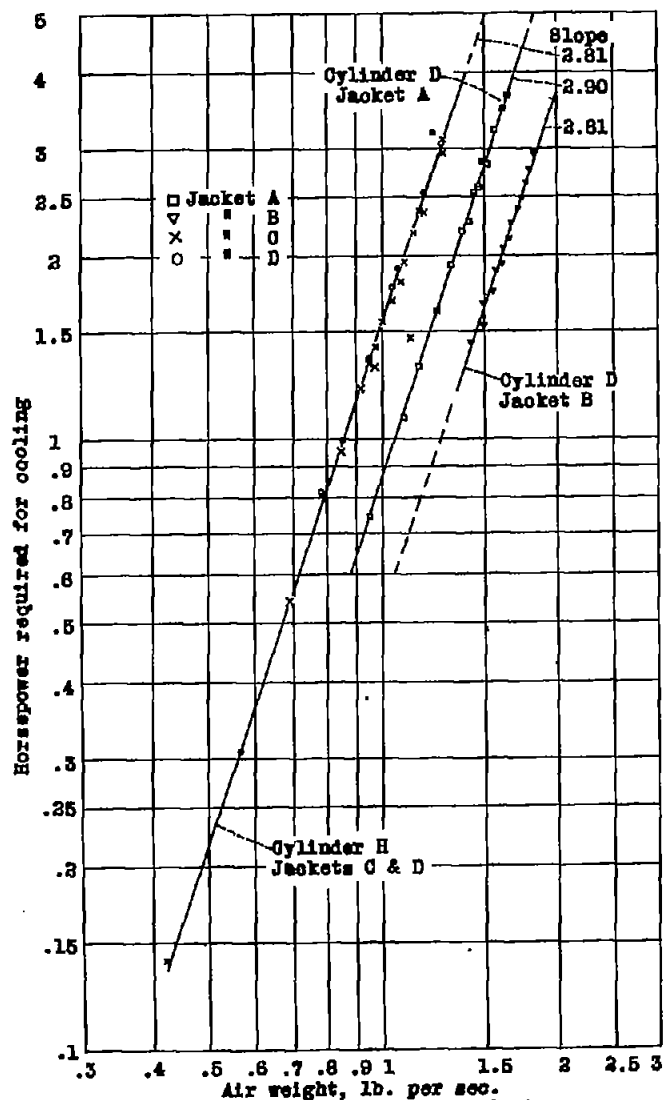


Figure 11.—Effect of cooling-air weight on power required for cooling. Blower efficiency, 100 percent.

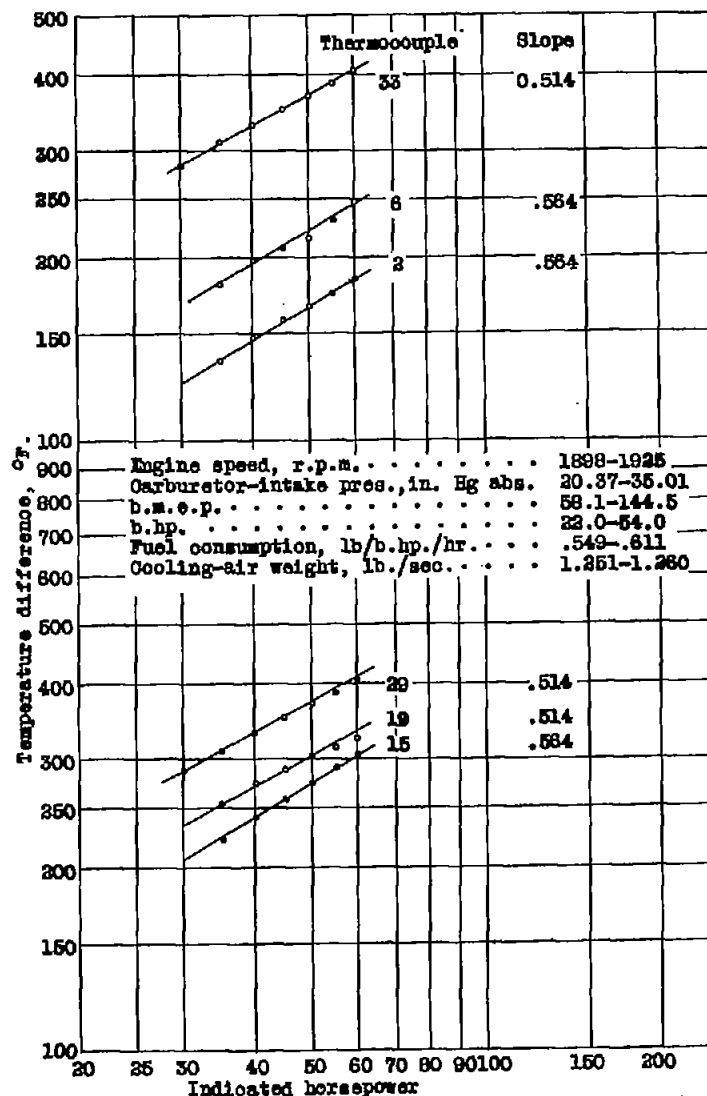


Figure 12.—Effect of indicated horsepower on temperature difference between the cylinder and the air for various thermocouples on cylinder H with jacket C.

